

Experimental and numerical flow investigation of Stirling engine regenerator



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ARTICLE INFO

Article history:

Received 10 August 2013

Received in revised form

13 May 2014

Accepted 1 June 2014

Available online 26 June 2014

Keywords:

Stirling engine regenerator

Test bench

Oscillating flow

Pressure drop

Thermal efficiency

Porous media

ABSTRACT

This paper presents both preliminary experimental and numerical studies of pressure drop and heat transfer characteristics of Stirling engine regenerators. A test bench is designed and manufactured for testing different regenerators under oscillating flow conditions, while three-dimensional (3-D) numerical simulations are performed to numerically characterize the pressure drop phenomena through a wound woven wire matrix regenerator under different porosity and flow boundary conditions.

The test bench operating condition range is initially determined based on the performance of the commercial, well-known Stirling engine called WhisperGen™. This oscillating flow test bench is essentially a symmetrical design, which allows two regenerator samples to be tested simultaneously under the same inflow conditions. The oscillating flow is generated by means of a linear motor which moves a piston in an oscillatory motion. Both the frequency and the stroke of the piston are modified to achieve different test conditions.

In the numerical study, use of a FVM (finite volume method) based CFD (computational fluid dynamics) approach for different configurations of small volume matrices leads to a derivation of a two-coefficient based friction factor correlation equation, which could be later implemented in an equivalent porous media with a confidence for future regenerator flow and heat transfer analysis.

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1. Introduction

Nowadays Europe faces changes, driven mainly by the economic crisis, globalization, demand on renewable resources, and ageing. Focusing on the demand on renewable resources, the priority is to promote a sustainable growth based on efficient technologies that can operate with different green energy sources; one of the most promising technologies is the Stirling engine [1,2]. The Stirling engines have suitable applications including conversion of solar energy, co-generation, submarine and space applications. Regarding micro-CHP (combined heat and power), the Stirling

engine WhisperGen™ is an example of high efficiency with very low levels of chemical and noise pollution.

The regenerator is usually the Stirling engine component that handles more power [3]. Therefore, the function of the regenerator is the key to the efficient performance of Stirling engines. Most of the research works in this field are carried out for the optimization of this component [4–6].

The optimization process for the Stirling engine regenerators mainly focus on increasing the heat transfer capacity and reducing pressure losses. Pressure losses through the regenerator have a direct influence on the Stirling engine indicated power and, consequently, on the electrical power output. Ibrahim and Tew [3] pointed out that a performance survey for small (<100 We) engines indicates that regenerator thermal inefficiency contributes 1.5% to engine thermal inefficiency while pressure drop losses contribute about 11% engine inefficiency. Therefore, the characterization of these phenomena through experimental, theoretical and numerical

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Nomenclature

A_{wr}	wetted area of regenerator, $A_{wr} = 4V_{dr}/d_h$, m^2	p	pressure, Pa
A_{xr}	cross-sectional area of regenerator, $A_{xr} = V_{dr}/L_r$, m^2	q	heat flux, W/m^2
a_1	form resistance parameter	R	specific gas constant, $J/(kg\ K)$
a_2	surface friction parameter	S_h	heat source term, W/m^3
C_f	friction factor = Darcy's friction factor = $f_D = \Delta p / (1/2 \rho u_{mx}^2 d_h / L)$	T_C	compression space temperature, K
c_p	specific heat of gas at constant pressure, $J/(kg\ K)$	T_E	expansion space temperature, K
c_r	specific heat of regenerator material, $J/(kg\ K)$	T_g	gas temperature, K
c_v	specific heat of gas at constant volume, $J/(kg\ K)$	T_{gL}	gas temperature at hot side, K
d_h	matrix hydraulic diameter, m	T_{g0}	gas temperature at cold side, K
d_w	wire diameter, m	T_{ref}	reference temperature at which physical properties are computed, K
E	energy per unit of mass, J/kg	T_{wr}	regenerator matrix temperature, K
h	convective heat transfer coefficient, $W/(m^2\ K)$	T_{wrL}	regenerator matrix temperature at hot side, K
K	permeability, m^2	T_{wr0}	regenerator matrix temperature at cold side, K
k	thermal conductivity of the gas, $W/(m\ K)$	t	time, s
k_r	thermal conductivity of the regenerator material, $W/(m\ K)$	u	gas velocity, m/s
L_r	regenerator length, $L_r = V_{dr}/A_{xr}$, m	$\bar{u}$	mean gas velocity = $\dot{m}/(\rho A_{xr})$, m/s
$\dot{m}$	mass flow rate, kg/s	u_{mx}	interstitial gas velocity inside the matrix, m/s
Fo	Fourier number = $4\alpha_r/(\bar{u} d_h) = 4\alpha_r/(d_h \sqrt{RT_{ref}}) \cdot N_{ma}^{-1}$	V_{dr}	regenerator dead volume, $V_{dr} = V_r \epsilon_{IV}$, m^3
Ma	Mach number = $\bar{u}/\sqrt{RT_{ref}} = \dot{m}/\sqrt{RT_{ref}}/(pA_{xr})$	V_r	regenerator total volume, m^3
Pr	Prandtl number = $\mu c_p/k$	x	longitudinal coordinate along the regenerator length, m
Re	Reynolds number = $\rho \bar{u} d_h/\mu = \dot{m} d_h/\mu A_{xr}$	α_r	thermal diffusivity of regenerator material, m^2/s
St	Stanton number = $h/(\rho \bar{u} c_p) = hA_{xr}/(\dot{m} c_p)$	ϵ	regenerator thermal efficiency
TCR	thermal capacity ratio = $\rho_r c_r/(\rho c_p) = (\rho_r c_r T_{ref}/p) \cdot (\gamma - 1)/\gamma$	γ	adiabatic coefficient
n_s	engine frequency, rev/s	μ	viscosity of gas, Pa s
		ϵ_{IV}	regenerator matrix volumetric porosity
		ρ	density of gas, kg/m^3
		ρ_r	density of regenerator material, kg/m^3

studies is crucial in co-generation applications to maximize the ratio between generated electrical power and source energy input.

Most of the research works mainly focus on the empirical characterization of the pressure losses for stacked woven wire regenerator matrices. Frequently, the phenomenon is modelled as a case of internal flow through a conduit, and the pressure drop is computed by means of a friction factor correlation based on the Hagen–Poiseuille's law. The correlation by Kays and London [7] is probably the most widely used. It assumes steady, incompressible flow and provides a friction factor, C_f correlation obtained from experiments performed in a perfectly pressed stacked woven wire screen matrices. Urieli and Berchowitz [8] proposed an equation based on Kay's and London's data. Seume and Simon [9] provided a review of the friction factor correlations for steady-state flow, and they also studied the compressibility effects and the characteristics of the oscillating flow in Stirling engine regenerators. Sodré and Parise [10] carried out experiments to determine the pressure drop through an annular conduit filled with a plain square wire-mesh woven-screen matrix. A corrected Ergun equation was used here to correlate the experimental data, considering the wall effects.

Several experimental test benches are used to characterize both pressure losses and convective heat transfer in regenerators under unidirectional, steady flow conditions [11]. However, testing facilities designed for oscillatory flow are more suitable for the operation conditions of Stirling regenerators. Wood et al. [12] used two different test benches, one for oscillating flow and other for steady unidirectional flow. Both test benches are designed for conventional heat exchangers, but they could be adapted for regenerators. The test bench used by Ibrahim et al. [13] consisted of an engine with two horizontal opposite pistons, crankshafts and individual drives which provide oscillating flow. Miyabe et al. [14], Tanaka et al. [6], Gedeon and Wood [15] also used experimental devices

under oscillating inflow conditions. The latter three studies [6,15] are classical references concerning friction factor and heat transfer experimental researches for stacked woven wire screens with variety of materials and geometries.

Experimental works are complemented with analytical and numerical approaches, usually related to the flow through wire screen matrices. The FVM (finite volume method) appears to be promising numerical discretization technique for the solution of governing partial differential equations as indicated by Rühlich and Quack [5], Gedeon and Wood [15], Ibrahim et al. [16], Tew [17] and others. These numerical studies suggest that the flow simulation is worthwhile to understand the flow behaviour and hence to characterize fluid flow friction for systems of potential regenerator applications.

Undoubtedly, the most widely used type of regenerator is the metal wire matrix, mainly stainless steel. In this approach, the regenerator volume is filled with wires or fibers of small diameter, typically from 40 to 150 μm (4×10^{-5} – 15×10^{-5} m). In the variant most likely used, the wires are previously woven into screen or meshes which are stacked adjoining to each other. There are different types of woven wire which lead to regenerators with different thermodynamic behaviour. Regenerators made of random-fibers or metal felts are cheaper to manufacture than stacked woven wire regenerators [15].

In the present study, a new test bench is designed for experimental characterization of the Stirling engine regenerator to be operated under oscillatory flow conditions. During the design, set-up, and preliminary tests of the test bench, a few limitations and measurement problems are identified. The test simplifications are realized to generalize the flow features and heat transfer characteristics however these also lead to large experimental errors. Therefore, a macro scale-finite volume method (FVM) based

numerical methodology is employed to overcome these limitations and/or problems to some extent and to propose more accurate preliminary flow solutions over a very large range of porosity and over a specified larger range of Reynolds number at different inflow velocities for a proper characterization of pressure drop phenomena in the Stirling engine regenerator.

2. Main physical phenomena in Stirling engine regenerators

The Stirling regenerator performance depends on geometrical parameters (d_h , η_v , A_{xr} , A_{wr} , L , V_{dr}), physical properties of the regenerator material (k_r , ρ_r , c_r), the working fluid properties (μ , c_p , γ , ρ , k), and operating variables ($\dot{m}$, n_s , T_E , T_C , p).

The main physical phenomena to be considered in the study of the Stirling regenerator are summarized in the sub-sequent sections with reference to research works undertaken in recent decades.

2.1. Pressure drop

The pressure decreases as the working gas circulates through the matrix of the regenerator, which in turn leads to an indicated power losses. The flow inside the matrix is modelled as a duct flow; in this case the pressure drop is obtained in analogy to Hagen–Poiseuille's law or to equivalent Darcy's law. Other researchers model the pressure losses in analogy to Forcheimer's law to take into account the pressure drop, not only caused by viscosity but also by the acceleration and deceleration effects of the fluid as it travels through the tortuous flow path of the porous media.

Miyabe [14] derived generalized experimental equations of flow friction factor for packed wire screens with variety of geometries. Tanaka [6] investigated the flow characteristics of the regenerator in an oscillating flow for stacked woven wire and felt type matrix. The prediction of pressure drop loss was possible by use of the hydraulic diameter as the representative length defined by the friction factor and Reynolds number. Gedeon and Wood [15] derived generic correlations for friction factor based on the test samples of a number of stacked woven wire and felt type matrix, with a range of porosities.

Following a similar method to that recently used in the experimental study of a non-tubular Stirling engine heater [18], theoretically the friction factor, C_f can be expressed for Stirling regenerators by means of the following functional relationship with seven dimensionless groups:

$$C_f = F \left(Re, Ma, \frac{L}{d_h}, \eta_v, \frac{\Delta T_g}{\bar{T}_{wr}}, \frac{\bar{T}_g}{\bar{T}_{wr}}, \frac{\Delta T_{wr}}{\bar{T}_{wr}} \right) \quad (1)$$

where $\bar{T}_g$ and $\bar{T}_{wr}$ denote, respectively, the mean temperatures of the gas and the regenerator matrix along the regenerator length, $\Delta T_g = T_{gL} - T_{g0}$ and $\Delta T_{wr} = T_{wrL} - T_{wr0}$.

The Mach number is included into Eq. (1) to account for possible compressibility effects, while the temperature ratios takes into account the spatial variations of physical properties of both the working gas as the regenerator material. In fact, the behaviour of the gas may be different during the heating and the cooling processes, but in practice it seems acceptable to obtain C_f from averages of measurements in both directions.

The Eq. (1) agrees with the experimental measurements carried out by Su [19] for steady, compressible flow through stacked screens, as well as with the analyses for Stirling regenerators introduced by Organ [20]. However, the effects of the seven dimensionless group of Eq. (1) are required to be analysed with the experimental results because some of them could be negligible as

occurred with the Ma in the experimental study of a non-tubular Stirling engine heater [18].

2.2. Convective heat transfer

The heat transfer correlations are to be carefully determined as the friction factor correlations by taking into account the hypotheses under which they are obtained. Miyabe [14] carried out theoretical and experimental studies on heat transfer for different types of wire mesh. Tanaka [6] analysed the regenerator behaviour under oscillating flow conditions, based on experimental data for metallic wire meshes, sintered metals and metal foams. More recently, Gedeon and Wood [15] performed an extensive experimental study of the regenerator under oscillating flow conditions, analysing wire mesh and felt-type regenerators.

A similar method to that recently used in the experimental study of a non-tubular Stirling engine heater [18] can also be applied to analyse the convective heat transfer coefficient in Stirling regenerators, which allows the following expression to be recommended as the basis for obtaining experimental correlations:

$$St = F \left(Re, Pr, Ma, \gamma, \frac{L}{d_h}, \eta_v, \frac{\Delta T_g}{\bar{T}_{wr}}, \frac{\bar{T}_g}{\bar{T}_{wr}}, \frac{\Delta T_{wr}}{\bar{T}_{wr}}, \frac{\rho_r c_r \Delta T_g}{p}, \frac{\alpha_r}{r_h \sqrt{R \Delta T_g}} \right) \quad (2)$$

2.3. Conduction heat transfer

The convective heat transfer between the working gas and the regenerator matrix is related to the heat conduction inside the regenerator material. This relation is represented in the Eq. (2) by the last two non-dimensional groups, which are proportional to the thermal capacity ratio TCR and the Fourier number Fo , respectively.

To study the radial heat conduction, theoretical methods are more common than experimental procedures [14].

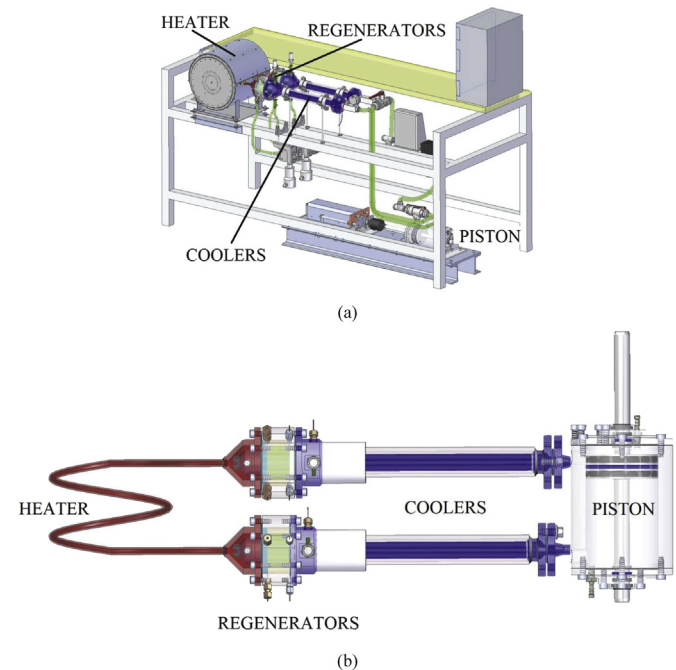


Fig. 1. A schematic of the SRTB configuration designed.

Table 1
Operating conditions of the micro-CHP Stirling engine WhisperGen™.

Properties	Operating conditions		
Pressure range	15–32 bar	Hot side temperature	550 °C
Maximum pressure	40 bar	Cold side temperature	80 °C
Working gas	Nitrogen	Maximum mass flow	0.02 kg/s
Heat exchanger power	3000 W	Maximum frequency	25 Hz

Table 2
Characteristics of the SRTB heat exchangers.

	Material	Number of tubes	Inner diameter	Length	Maximum power
Electrical heater	Stainless steel	1	0.01 m	0.94 m	5400 W
Water coolers	Aluminium	3	0.01 m	0.35 m	2200 W

For the investigated Reynolds number range of 60–350, the flow regime can be considered to be laminar [9] in the regenerators of benchmark prototypes.

3. Experimental study

3.1. Test bench configuration

The SRTB (Stirling Test Bench Regenerator) is designed in accordance with the configuration used by Tanaka et al. [6] by replacing the piston drive with a linear actuator, in such a way that the cylinder can be completely sealed and can facilitate testing the flow at high pressures. The SRTB is essentially symmetrical device which is comprised of a central heater, two regenerators, two coolers and a piston as shown in Fig. 1. It is intended to obtain a practically steady oscillating flow in both directions of piston stroke. This arrangement avoids the use of high-temperature-resistant seals because the piston is operating at lower temperatures and allows the two regenerators to be tested simultaneously under the same flow conditions, which is important for calibration purposes.

The dimensioning of SRTB is realized considering the operating range of the micro-CHP Stirling engine WhisperGen™ (Table 1).

The linear electric actuator has a 150 mm stroke and is capable of exerting a continuous force of 2400 N while providing a

maximum speed of 0.953 m/s. The main particulars of the tubular heat exchangers arranged in the SRTB are summarized in Table 2.

Two regenerator specimens can be accommodated for a simultaneous testing by means of two sample holders, formed mainly by a support part and a conical taper part of stainless steel (Fig. 2). The function of the taper part is to direct and uniform outflow of the heat exchangers into the matrix of the regenerator. The matrix is thermally isolated from the sample holder by means of mica which thermal conductivity is equal to 0.35 W/(m K). The gas temperature is measured at both input and output sections of each regenerator. The matrix temperature is measured at three points separated from each other by 120°, both in the input section and in the output of each regenerator.

3.2. Data acquisition system

In the SRTB, the following variables are monitored by means of transducers whose characteristics are summarized in Table 3: T_{gl} , T_{go} , T_{wrl} , T_{wr0} , Δp , p , $\dot{m}$ and linear actuator position.

3.3. Preliminary test results

In the micro-CHP Stirling engine WhisperGen™, each piston has a stroke of 0.025 m and reaches a maximum velocity of 1.25 m/s, i.e. an operating frequency of 25 Hz, which is approximately 20% higher than the maximum frequency provided by the SRTB linear actuator (19 Hz). However, the frequency of the fluid oscillation is limited by the highest transducer response time, which corresponds to the MFM (mass flow meter).

The MFM yields a direct mass flow measurement based on the Coriolis force principle and it is designed for measuring a one-way steady flow with a settling time of 0.5 s. The LA (linear actuator) moves in a reciprocating motion at constant speed in each direction, being a triangular-wave of position vs time, so it produces a square-wave mass flow. During the negative half of the mass wave, the MFM is trying to measure a reversed mass flow, so it does not produce any significant measurement at all. When the square-wave mass flow switches from negative to positive value, the MFM starts to measure and it takes a time equal to the settling time of 0.5 s to produce a significant measurement. This settling time limits the maximum frequency of the LA and mass flow to 1 Hz. Because of this, most of the time the MFM is working in a transient regime and only after the settling time elapses in each positive half-wave, it yields valid measurements. To overcome this drawback, it is

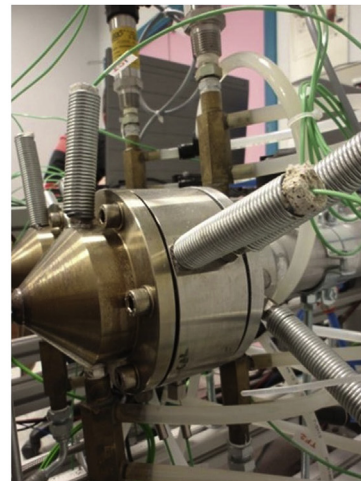
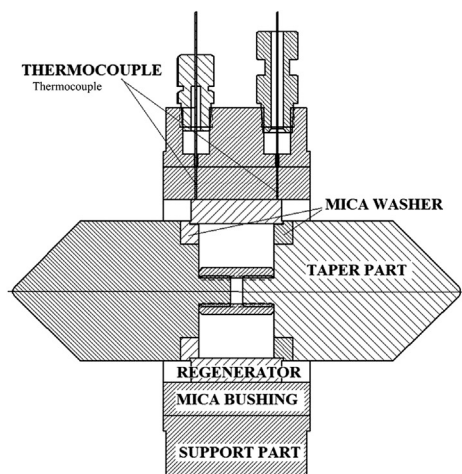


Fig. 2. Sample holder details.

Table 3
Characteristics of the SRTB transducers.

Transducer	Range	Response time	Accuracy	Quantity
Mass flow meter	0.4–40 g/s	N/A	1.0%	1
Pressure transducer	0–40 bar	5 ms	0.2%	2
Differential pressure transducer	–200 to 200 mbar	5 ms	0.1%	2
Thermocouple (Type K)	up to 1100 °C	50 ms	0.5%	16

assumed that after the settling time elapses, there is an acquisition time interval in which the measurements produced by the MFM are inside an error band of 3% of the maximum mass flow rate (Fig. 3). Therefore, the valid value of the mass flow rate is assumed to be average of the measured values that are greater than 97% of the maximum values measured during the acquisition time interval.

Due to above summarized limitations, obtaining high Re values to construct correlations based on Eq. (1) or (2) requires that the pressure be regulated adequately. The limitation is greater for Ma , which can only be regulated by means of the linear actuator frequency. Thus, in the current SRTB version, the mass flow rate should be calculated from indirect measurements when tests for a wide range of variables are necessary.

Fig. 4 shows the variation of the instantaneous pressure drop test results obtained for a regenerator at gauge pressure of 16.2 bar, mass flow rate of 0.0086 kg/s and actuator velocity of 0.138 m/s. Fig. 4 also shows the instantaneous position of the linear actuator and the mass flow rate recorded with a settling time higher than the other two parameters (see Fig. 3). The time history of the instantaneous pressure drop changes in a triangular shape and a phase lag between the pressure drop and the linear actuator position is observed. The measured pressure drop is found to be higher in the cold blow time. This result is expected because the pressure drop is function of the Reynolds number which becomes larger under constant working pressure and velocity during the cooling process due to increasing density.

Although the following equations represents a very simplified form of the general energy conservation equation to construct a basis for a comprehensive study of the regenerator applications, it may facilitate a reasonable qualitative interpretation of the present experimental results here for initial design purposes.

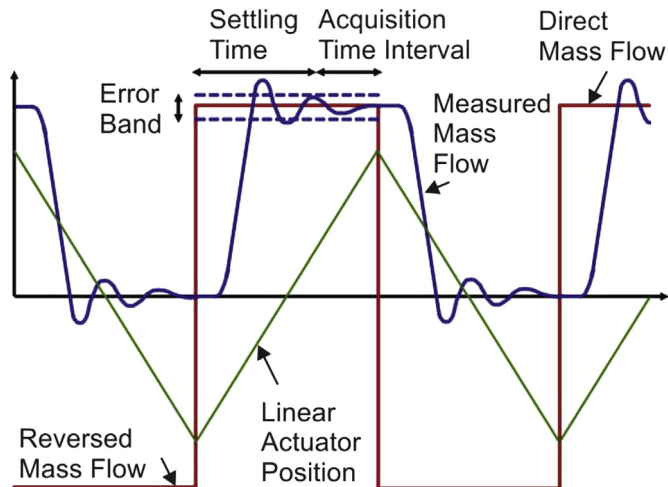


Fig. 3. Sequence of mass flow measurement.

Fig. 5 shows a volume of the regenerator matrix of hydraulic diameter d_h and length L_r , volumetric porosity ϵ_v , temperature of the solid matrix wall T_{wr} , temperature of the working gas T_g and a differential control volume $A_x dx$. The solid matrix to gas heat transfer Q can be expressed in terms of average heat transfer h coefficient for the whole regenerator volume $A_x L_r$ as follows:

$$Q = hA_{wr}(T_{wr} - T_g) \quad (3)$$

Starting from one-dimensional gas energy equation given by Eq. (4):

$$\frac{\partial}{\partial t}(\rho E) + \frac{\partial}{\partial x}[u(\rho E + p) + q] - S_h = 0 \quad (4)$$

Considering the kinetic energy effects neglected ($u^2/2$) in Eq. (4), and expressing $\rho E = (c_v/R)p$ and $\rho E + p = c_p RT_g$, the alternative form of the energy equation is:

$$\frac{c_v}{R} \frac{\partial p}{\partial t} + \frac{\partial}{\partial x}[\rho u c_p T_g + q] - S_h = 0 \quad (5)$$

Eq. (5) is transformed into Eq. (6) assuming the mass flow is uniform in the matrix, the specific gas capacities are independent on temperature, the matrix temperature is linear and the apparent conductivity is not a function of x . By combining the last two assumptions, the term becomes as $(\partial q/\partial x) = k_a(\partial^2 T/\partial x^2) \approx 0$. In the differential control volume $A_x dx$ the gas mass is the density ρ multiplied by the fraction of the control volume occupied by the gas, $\epsilon_v A_x dx = A_{xr} dx$ and considering the solid matrix to gas heat transfer the gas energy equation is converted to:

$$\left[\rho c_v \frac{\partial T_g}{\partial t} + \rho u c_p \frac{\partial T_g}{\partial x} \right] A_{xr} dx = hA_{wr}(T_{wr} - T_g) \frac{dx}{L_r} \quad (6)$$

The gas energy Eq. (6) is expressed in terms of regenerator matrix hydraulic diameter as follows:

$$\rho c_v \frac{\partial T_g}{\partial t} + \rho u c_p \frac{\partial T_g}{\partial x} = \frac{4}{d_h} h(T_{wr} - T_g) \quad (7)$$

For the regenerator matrix, the solid energy equation of the control volume is given by Eq. (8):

$$dm_r c_r \frac{\partial T_{wr}}{\partial t} = hA_{wr}(T_g - T_{wr}) \frac{dx}{L_r} \quad (8)$$

The elemental regenerator matrix mass, dm_r is obtained multiplied the regenerator material density ρ_r by the fraction of the control volume occupied by solid material. In terms the volumetric porosity the fraction is $A_{xr}(1 - \epsilon_v) dx/\epsilon_v$. Therefore, the regenerator solid energy equation is transformed into Eq. (9):

$$\rho_r c_r A_{xr} dx \frac{(1 - \epsilon_v)}{\epsilon_v} \frac{\partial T_{wr}}{\partial t} = hA_{wr}(T_g - T_{wr}) \frac{dx}{L_r} \quad (9)$$

The solid energy Eq. (9) is expressed in terms of regenerator matrix hydraulic diameter as follows:

$$\rho_r c_r \frac{\partial T_{wr}}{\partial t} = \frac{4}{d_h} \frac{\epsilon_v}{(1 - \epsilon_v)} h(T_g - T_{wr}) \quad (10)$$

Fig. 6 shows a conceptual graphical representation of Eq. (6) for the temperature variations as function of the position and time, assuming that the regenerator temperature is independent of time at the wire surface, i.e. $T_{wr} \approx T_{wr}(x)$. In Fig. 6 at a time instant labelled as (1), the gas circulates from the hot to the cold end ($u < 0$), since $\partial T_{wr}/\partial x > 0$, the gas leaves heat to the regenerator matrix ($T_{gx} > T_{wrx}$) and the gas temperature is at a maximum value ($\partial T_{gx}/\partial t = 0$). The gas also flows from the hot to the cold end at a

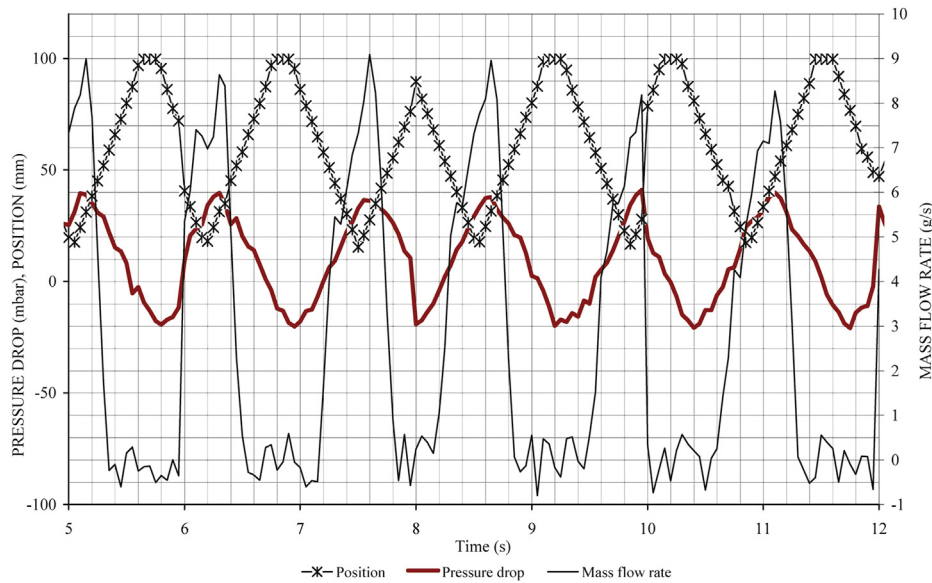


Fig. 4. Example of pressure drop test results.

time labelled as (4), since $T_{gx} = T_{wrx}$ and $\partial T_{gx}/\partial t > 0$. However, it circulates in the opposite direction ($u > 0$) in the time labelled as (2) ($T_{gx} = T_{wrx}$; $\partial T_{gx}/\partial t < 0$) and as (3) ($T_{gx} < T_{wrx}$; $\partial T_{gx}/\partial t = 0$). The flow direction is reversed in some intermediate time labelled as (0).

Preliminary tests show the difficulty of measuring the temperature of the regenerator matrix, since it is observed that some thermocouples frequently give records close to the gas temperature values due to an insufficient contact with the matrix. In attempting to solve this problem it is found that the temperature of the regenerator matrix substantially coincides with the time average of the gas temperature at the corresponding end, in accordance with the scheme of the Fig. 6. Consequently, it seems that temperature measurements of the regenerator matrix are not necessary, and this significantly reduces the cost of the tests associated with the correct

placement of the thermocouples. Fig. 7 shows the temperature results measured in the same regenerator and conditions of Fig. 4.

It may be noticed from Figs. 4 and 7 that the valid mass flow rate data are of the peak values measured during the very small acquisition time interval that occurs after the settling time and near to the reversal of the flow, which correspond approximately to the points labelled as (0) in Fig. 6. On the other hand, the SRTB function consists not only of providing facilities to obtain correlations for a given regenerator type, but also of performing preliminary tests to make comparisons among different regenerator types. For this second utility, the concept of regenerator thermal efficiency is of interest.

By taking into account of the typical temperature variations of the oscillating flow through regenerators, it is considered adequate basing the thermal comparisons between regenerators on the

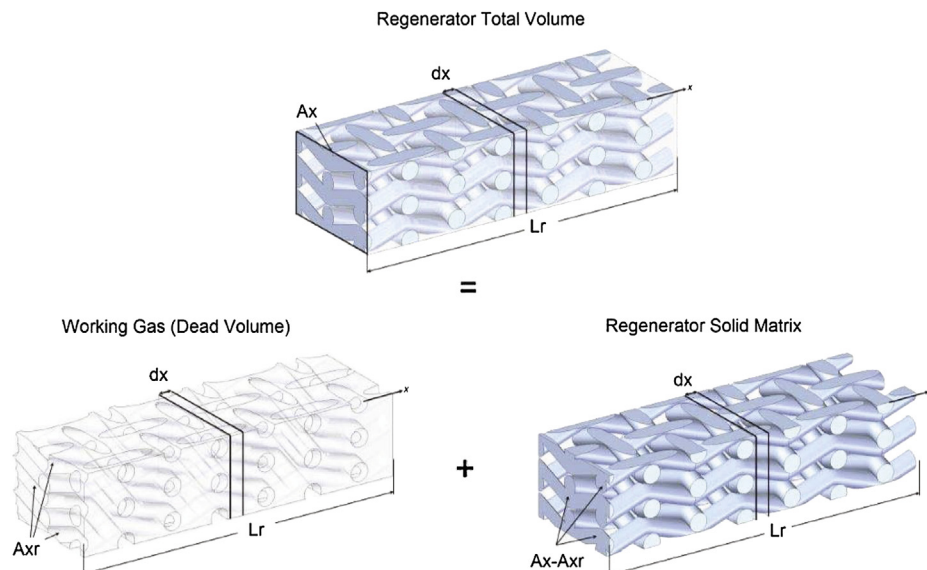


Fig. 5. Thermal analysis of regenerator.

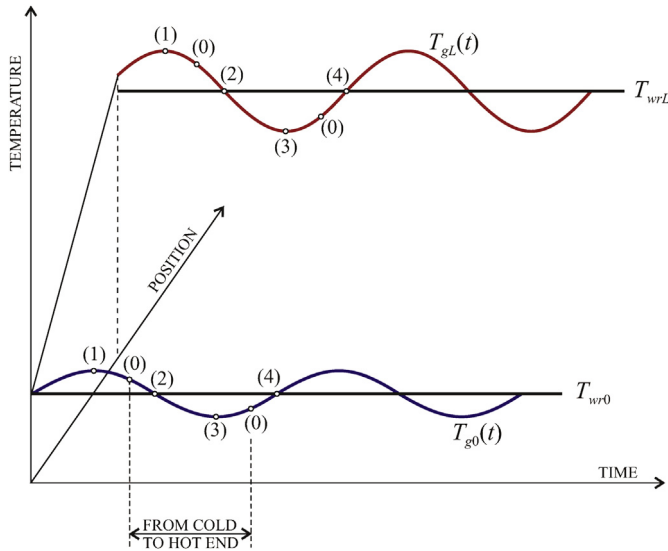


Fig. 6. Conceptual scheme of flow directions and temperature distributions.

following definitions of the heating and cooling thermal efficiencies:

$$\varepsilon^+ = \frac{T_{gL,min} - T_{g0,min}}{\bar{T}_{wrL} - T_{g0,min}} \quad (11)$$

$$\varepsilon^- = \frac{T_{gL,max} - T_{g0,max}}{T_{gL,max} - \bar{T}_{wr0}} \quad (12)$$

where the subscripts max and min refer to the maximum and minimum values measured by the thermocouples at the corresponding sections, while $\bar{T}_{wr0}$ and $\bar{T}_{wrL}$ denote the time averages of the regenerator matrix temperatures at each section. In order to diminish the noise effects on the measurements, the 97% and 3% are used to estimate those maximum and minimum values.

Table 4

The computed efficiencies for two regenerators of 110 μm wire diameter at the pressure of 10 bar.

	10%		25%		40%	
	Reg. 1	Reg. 2	Reg. 1	Reg. 2	Reg. 1	Reg. 2
ε^+	91.3%	89.9%	94.9%	94.0%	95.4%	94.6%
ε^-	99.0%	98.1%	99.0%	98.7%	99.0%	98.3%

Table 4 allows the comparison between the heating and cooling efficiencies obtained in simultaneous measurements for two regenerators of 110 μm wire diameter, at the pressure of 10 bar and 10% of the maximum actuator velocity, which lead to a mass flow rate of 0.0023 kg/s at the percentile of 97%. The results obtained for two higher velocities (25% and 40%) are also shown.

It is observed here that the efficiency during the heating process increases in both regenerators with increasing mass flow rate while pressure increases. However, a clear influence of the mass flow rate over the efficiency during the cooling process is not observed. On the other hand, the efficiency is higher during the cooling process than during the heating process in all cases, which is consistent with the smaller amplitude of thermal oscillations as observed in Fig. 7 at the regenerator cold end. This is probably due to the fact that the efficiency is defined in terms of temperature differences here. The mass flow rate is varied by controlling the gas pressure and velocity of the actuator. For a constant working pressure, the gas density is larger at the cold end of the regenerator due to lower temperature there.

4. Numerical study

The lack of robustness and limitations due to flow measurement and testing conditions, suggest that the numerical computations are required for a proper characterization of pressure drop in the Stirling engine regenerator. The preliminary simulations are therefore carried out for a specified range of Reynolds number and volumetric porosity level in association with a simplified equivalent porosity media model to initially derive a two-coefficient based friction correlation equation to be used with a confidence for a

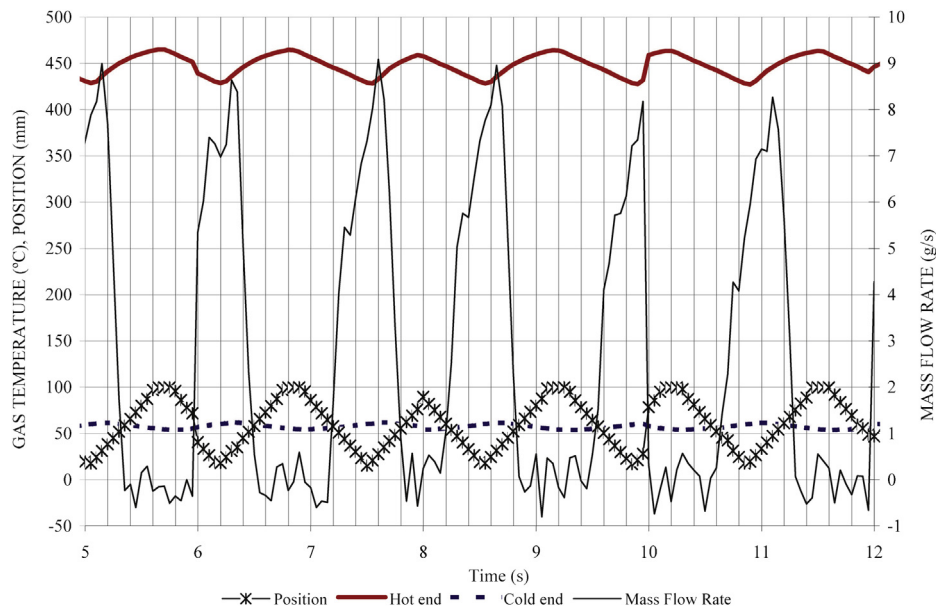


Fig. 7. Example of gas temperature measurements.

specified parameter range. A macro scale CFD (computational fluid dynamics) modelling approach is utilized with the preliminary objective of comparing the numerical simulation results against experimental test results for pressure-drop through the internal structure of a wound woven wire mesh matrix. By taking into consideration of the complexity of the flow inside the regenerator, the aim of the initial CFD simulations here is to study one of the main physical phenomena in the Stirling regenerator, the pressure drop. The computational modelling of the Stirling regenerator pressure drop losses is conducted with isothermal flow assumption and only one wire diameter, therefore the numbers of dimensionless group of Eq. (1) is drastically reduced. In the future, it is planned to extend the procedure to analyse the regenerator pressure drop characteristics together with the convective heat transfer.

4.1. Governing equations

In the present numerical study, the fluid is considered to be incompressible and Newtonian, with constant properties, and the flow is assumed to be three-dimensional (3-D), laminar, isothermal and steady. Then the 3-D laminar form of the steady Navier–Stokes equations is solved for the flow field. No energy equation is solved for the temperature field due to isothermal flow assumption. The governing continuity, and momentum equations can be expressed in tensor notation as:

$$\frac{\partial u_i}{\partial x_i} = 0 \quad (13)$$

$$\frac{\partial u_i u_j}{\partial x_j} = -\frac{1}{\rho} \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\nu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] - g_i \quad (14)$$

where u_i and p are fluid velocity and pressure, respectively while ρ and μ are the density and dynamic viscosity of the fluid, respectively.

4.2. Numerical solver

The governing flow equation namely continuity equations and Navier–Stokes equations with the associated boundary conditions imposed are solved numerically on the basis of viscous, incompressible, Newtonian and laminar fluid flow assumptions in 3-D manner. All fluid properties (including density, viscosity, specific heat, and conductivity) are assumed constant. The governing equations are discretized and solved sequentially using a finite volume method (FVM) based numerical flow solver [21,22] with a second order upwind scheme for the discretization of the continuity and momentum equations for the laminar flow solutions. The convergence criterion for all the velocity components and for the continuity is set to 10^{-6} for all simulations.

4.3. Model geometry and computational domain

The numerical models are developed taking as a reference the Stirling regenerator matrix of the WhisperGen™ micro-CHP Stirling engine. Table 5 summarizes the main features of wound woven wire mesh regenerator matrix used.

Table 5
Main mesh specifications.

Wire diameter	Mesh opening	Mesh thickness	Misalignment	Porosity
80 μm	140 μm	160 μm	Random	~60%

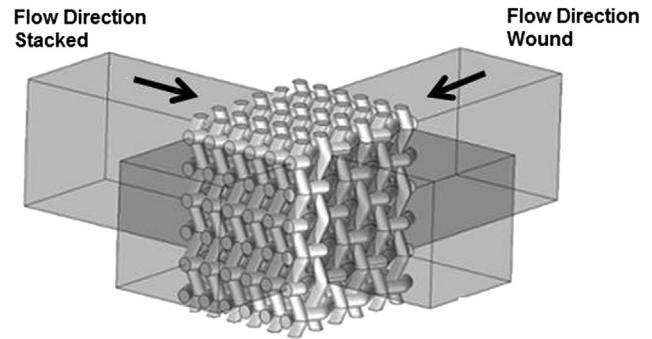


Fig. 8. Woven wire mesh flow direction: stacked and wound.

Fig. 8 illustrates the flow direction difference between a stacked woven wire mesh and a wound woven wire mesh configuration.

Macroscopic measurements of the wound woven wire mesh matrix are conducted in order to verify the actual internal geometry of the matrix; it is observed that the average separation between mesh layers is higher than the theoretically calculated value. In addition, there is an expected “random” misalignment between the wound mesh layers as observed in Fig. 9.

In the numerical study, the effect of porosity on the flow characterization in the Stirling regenerator model is investigated through a set of numerical simulations performed for different porosity level ranging from 52% to 72%, which may practically cover the matrix porosity range, as summarized in Table 6, with the same wire diameter and mesh geometry.

Since modelling the complete regenerator matrix is too costly in terms of equipment and computational time, a small matrix domain is generated as a representative of the whole Stirling regenerator engine in the early stage of the numerical procedure. The generated 3-D geometry domain is composed of five layers overlapping wound mesh as shown in Fig. 10, and the entrance area is function of the volumetric porosity of each simulation case and the length of mesh is 0.001 m.

As explained before, the aim is to study the behaviour of the working gas flow through the control volume. Therefore, the control volume of simulation case is generated to include the fluid volume surrounding the wires (not the wire material) with an associated input distance of 0.0005 m and an output distance of 0.0015 m to avoid backflow and to ensure that the flow is fully developed (Fig. 11(a) and (b)). The control volume mesh is composed of a hybrid mesh system, in which hexahedron mesh cells are utilized in the close vicinity of the entrance and exit regions, while tetrahedron mesh cells are utilized around the matrix of the mesh, with approximately 1,200,000 cells. For each simulation case, the maximum aspect ratio is below 2.9 and the maximum skewness is of 0.86.

4.4. Boundary conditions

All model simulations are carried out by considering the following below boundary conditions as illustrated in Fig. 12:

1. Inflow boundary: Velocity inlet boundary condition is used to define the uniform fluid velocity profile at the inflow boundary.
2. Outlet boundary: Pressure outlet boundary conditions are used to define the static (gauge) pressure at the outlet boundary and to eliminate reverse flow problem.
3. Side boundaries: Symmetry boundary conditions are used. Free-slip boundary conditions at the four side boundaries of the computational domain are imposed. The normal velocity

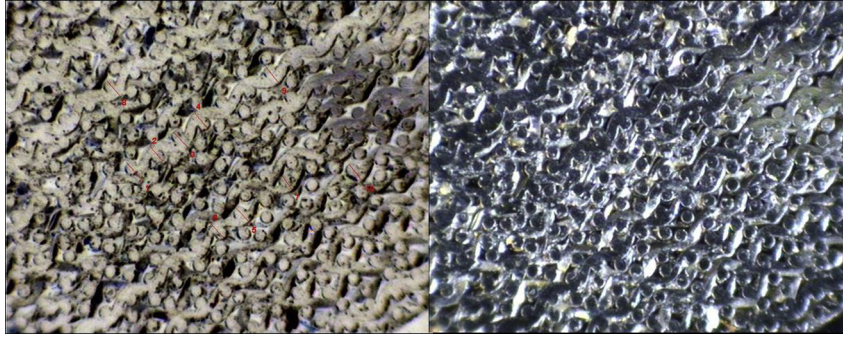


Fig. 9. Macroscopic picture of the wound woven wire mesh matrix.

components and the normal gradients of all velocity components are assumed to have a zero value.

4. Wall surfaces: No-slip wall boundary conditions together with the standard wall functions are used to define the interior wall boundaries between wires and flow for turbulent simulation cases.

In the present preliminary computations, the CFD methodology adopted is restricted to isothermal, steady flow performance, covering the Reynolds number range representative of the instantaneous values expected during the cyclic engine operation and one wound wire mesh (one diameter).

4.5. Equivalent porous media model

Once the initial simulations are performed for the whole geometry of the wire mesh matrix, i.e. full scale computations, an equivalent porous media model is incorporated in the present numerical methodology to simulate the full regenerator flow behaviour for future applications with inclusion of heat transfer phenomena. This full regenerator model can be based on the pressure drop results obtained for each full scale computation.

The geometry assumed for the equivalent porous model differs from that of the local detailed models as the new local mesh domain is modelled as an equivalent porous material. The rest of the control volume remains the same and the boundary conditions and simulation options are not modified.

Porous media can be modelled in the present fluid flow solver [22] with inclusion of a momentum source term to the standard fluid flow equations. The source term can be composed of two parts namely a viscous loss term, as in Darcy's law, and an inertial loss term, as in Forcheimer's law. This second term dominates at high velocities. Laminar, transient and turbulent flow are observed in regenerators of benchmark Stirling engines [9]. However, at first approach, the flow through the regenerator wire mesh matrix is expected to be laminar, and Darcy's law is used to determine the equivalent porous media model, according to the following equation:

$$\Delta p = -\frac{\mu}{K} u \Delta x \quad (15)$$

The values of permeability coefficient K is required to be define the equivalent porous media model are obtained from pressure drop values obtained from the numerical simulations performed at

different velocities for the local detailed mesh models, wherein the thickness of the porous medium, Δx has a value of 1 mm.

The numerical results can be used to represent a dimensionless chart of friction factor as a function of Reynolds number which is defined considering in terms of the matrix flow velocity and the matrix hydraulic diameter as follows:

$$Re_{mx} = \frac{\rho u_{mx} d_h}{\mu} \quad (16)$$

$$u_{mx} = \frac{\bar{u}}{\eta_v} \quad (17)$$

$$d_h = d_w \frac{\eta_v}{1 - \eta_v} \quad (18)$$

Moreover, Darcy–Weisbach's equation allows Darcy's friction factor to be introduced to express the dimensionless pressure drop as the following variations of the Euler number referred to the matrix flow velocity:

$$\frac{\Delta p}{1/2 \rho u_{mx}^2} = C_f \frac{L}{d_h} \quad (19)$$

In short, it is expected that the equivalent porous media model lead to numerical results in agreement with the following expression, which is a particular case of the Eq. (1) for unidirectional, incompressible, and isothermal steady flow conditions:

$$C_f = F(Re, \eta_v) \quad (20)$$

The predominance of viscous or inertial effects can be assessed using correlations with the following form of Ergun's equation, where the two adjusting parameters should depend on the porosity:

$$C_f = \frac{a_1}{Re} + a_2 \quad (21)$$

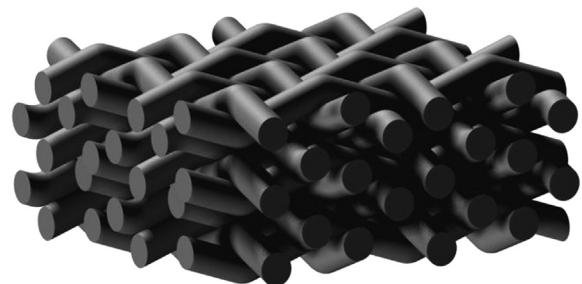
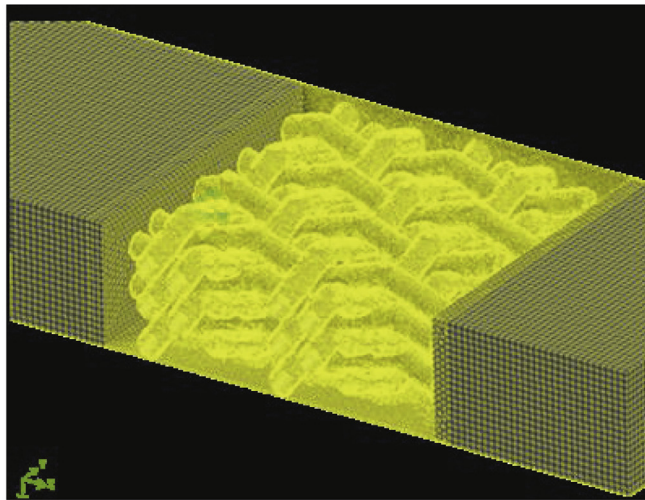


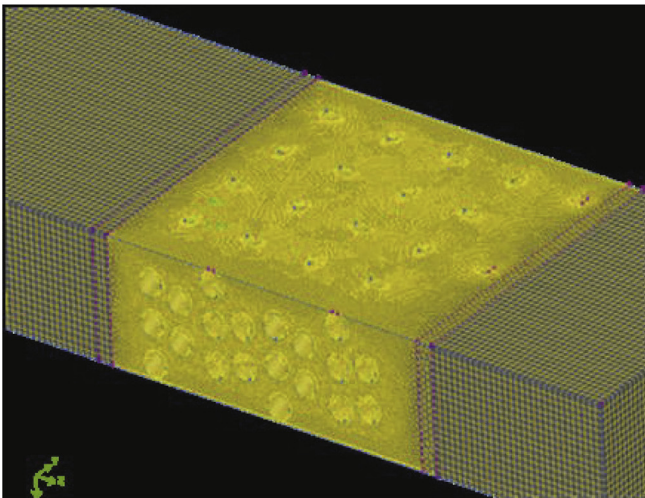
Fig. 10. Modelled geometry of wound woven wire mesh matrix.

Table 6
Porosity values assumed in numerical models.

Model number	1	2	3	4
Porosity	52%	60%	64%	72%



(a)



(b)

Fig. 11. Mesh: a) wire surface mesh and b) control volume mesh.

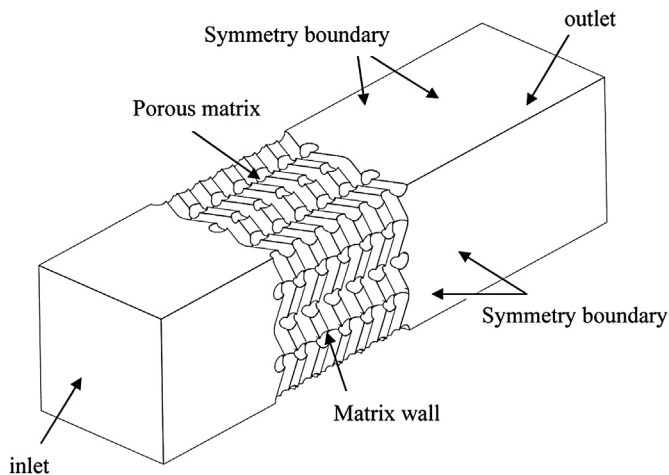
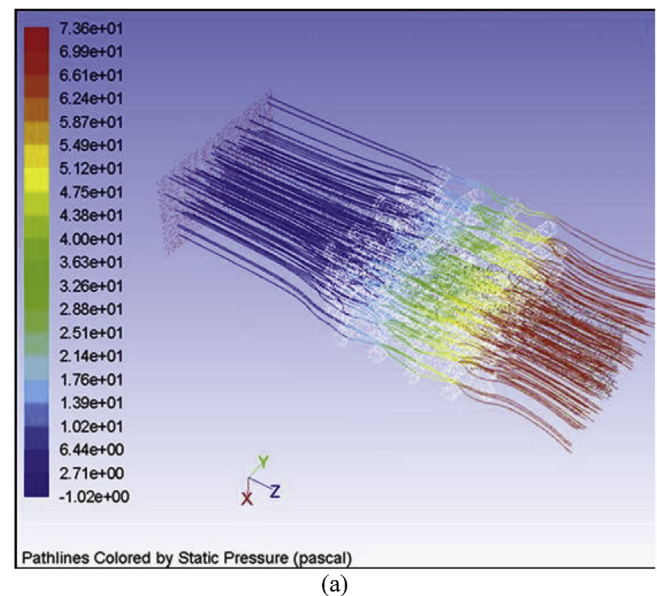


Fig. 12. Boundary conditions imposed for the present numerical model.

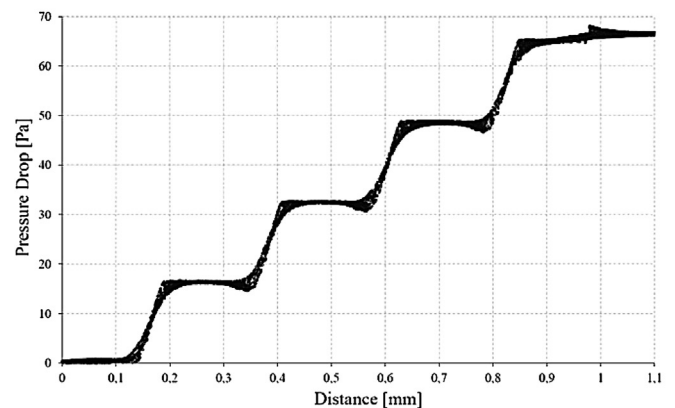
4.6. Numerical results

The numerical simulations are performed for a porosity ranging from 52% to 72% and for Reynolds number ranging from 5 to 50 in the laminar flow regime to investigate the effects of porosity and Reynolds number on pressure drop. A two-coefficient based friction correlation equation is numerically obtained and its validity for the specified porosity and Reynolds number range is assessed in comparison with the experimental data available in the literature. The numerical results are presented in Figs. 13–17.

Fig. 13 shows both quantitative and qualitative based pressure drop results obtained from the 3-D wire mesh matrix model with a matrix porosity level of 72%. It is clearly observed from the flow path lines coloured by the static pressure in Fig. 13a that the pressure drop occurs through the wire matrix. The pressure drop as a function of the length through the regenerator matrix is also quantitatively observed in Fig. 13b that the pressure drop behaviour is linear, stepwise and uniform through the regenerator matrix. In Fig. 13b the constant pressure drop corresponds to one obtained within the space between transversal woven wires and it increases when the flow passes through the transversal woven wires. This performance mainly depends on the geometrical configuration of



(a)



(b)

Fig. 13. Detailed model porosity 72% and 0.72 m/s: a) pathlines coloured by static pressure and b) pressure drop vs. distance. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

the woven wire matrix. In the model of 72% volumetric porosity, the pressure drop is approximately 18 Pa per layer with a total pressure drop of 68.9 Pa (4 layers) through the model for a distance of 1 mm. The results obtained from the rest of the matrix configurations with different volumetric porosity levels also reproduce the similar evolution behaviour of the pressure drop.

The plot of the pressure drop as function of the velocity through the different regenerator matrix configurations is also given in Fig. 14. It is observed here that the trend-line is approximated to a straight line in regards to the velocity, which corresponds to Darcy's law (Eq. (15)) for laminar flow in the velocity range investigated. It is observed an inverse influence of the volumetric porosity over the pressure drop, this means that the highest pressure drop is obtained for the volumetric porosity of 52% while the lowest pressure drop is obtained for the volumetric porosity of 72%. These results are expected considering that the volumetric porosity reduces the maximum velocity through the porous media (Eq. (17)) while increasing the hydraulic diameter (Eq. (18)), which in turn diminishes the pressure drop (Eq. (19)). However, the influence of the volumetric porosity over the friction factor needs to be further analysed.

These numerical results can be extrapolated to obtain the permeability coefficients for the equivalent porous media defined by Eq. (15). It should be noted here that all numerical results are obtained under constant working fluid properties of density and viscosity.

Fig. 15 shows the path lines obtained from the 72% volumetric porosity equivalent porous media model at an inlet velocity of 0.72 m/s, where the colour scale indicates the pressure drop through the matrix. The results for the equivalent porous media model based on the numerically obtained permeability coefficients show a good agreement with the results of the 3-D detailed wire mesh matrix model (see Fig. 13a), with a deviation of less than 2% for each model studied.

The combined effects of volumetric porosity and the Reynolds number on pressure drop are illustrated in Fig. 16 in which the pressure drop variation is evaluated in terms of the inflow velocity and volumetric porosity. This figure is a 3-D representation of the numerical data presented in Fig. 14, where the influence of the volumetric porosity and the velocity over the pressure drop is observed.

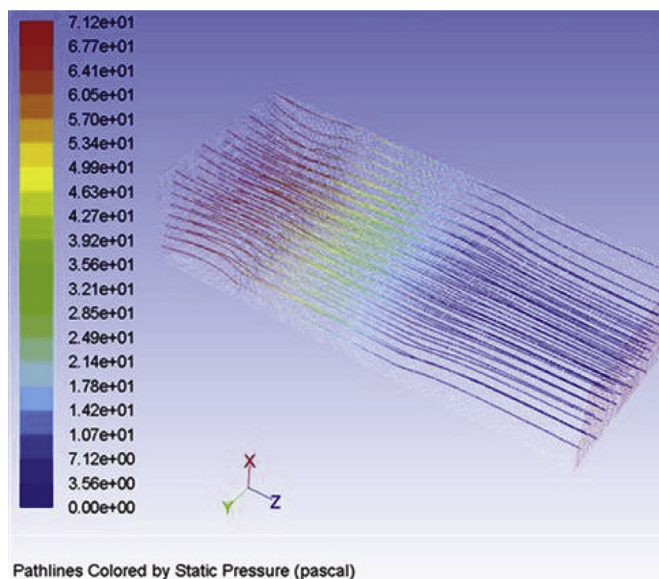


Fig. 15. Equivalent model porosity 72% and 0.72 m/s: pathlines colored by static pressure. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

Fig. 17 shows the friction factors, which are represented as a function of Reynolds number for four volumetric porosities studied. The Reynolds number is calculated based on the hydraulic diameter (Eq. (16)) while the friction factor is retrieved from the obtained pressure drop data (Eq. (19)). In the range of Reynolds number investigated, it is observed that the friction factor obtained from the volumetric porosity of 72% is the highest, followed by the matrix with a volumetric porosity of 64%. On the other hand, there are no significant variations in the friction factor values obtained for 52% and 60% volumetric porosities. Therefore, based on the present results it cannot be affirmed that there is a direct influence of the volumetric porosity over the friction factor. However, as previously shown in Fig. 14, there is a clear influence of the volumetric porosity on the pressure drop. Therefore, this initial parametric study should be extended to a wide range of volumetric porosities and different

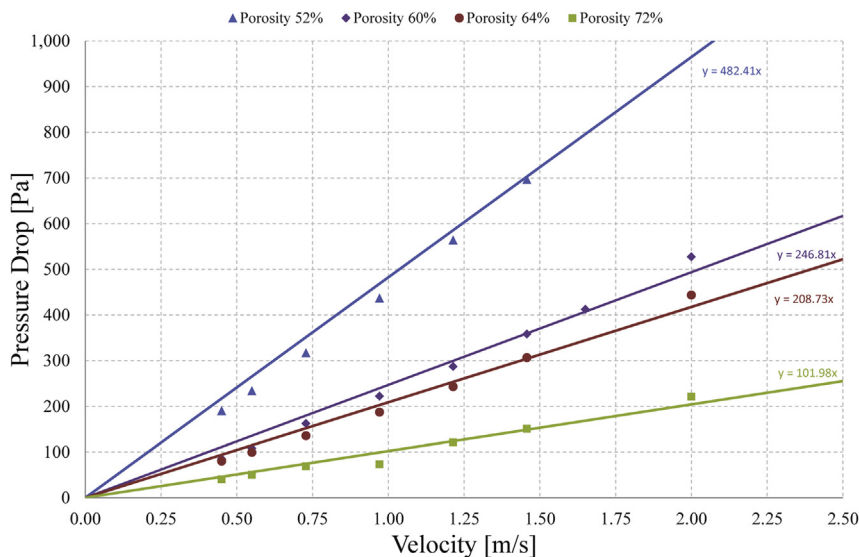


Fig. 14. Pressure drop versus matrix velocity at different volumetric porosity.

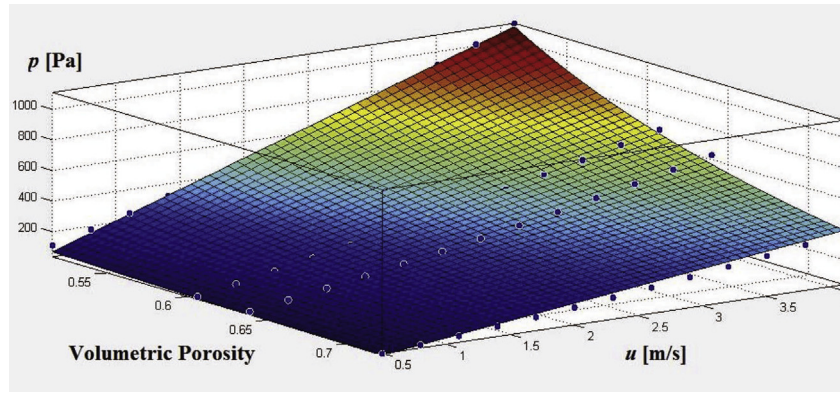


Fig. 16. Pressure drop versus matrix velocity and volumetric porosity.

woven wire mesh configurations in order to further analyse the influence of volumetric porosity over the pressure drop and friction factor correlation equation.

Finally, a preliminary two-coefficient based friction factor correlation equation is numerically derived in accordance with the Ergun equation (Eq. (21)) in terms of the Reynolds number. This equation, which is also utilized in Fig. 17 in accordance with the functional relation given in Eq. (21), can be defined as follows:

$$C_f = \frac{230}{Re} + 2.6 \quad (22)$$

Though the present numerical methodology in association with the equivalent porous media model does not completely represent the regenerator characterization for a very large range of parametric values for regenerator characterization, it gives a good indication of determining pressure drop in the regenerator and hence providing a friction correlation equation to be used in confidence for a specified range of Reynolds number and volumetric porosity level.

5. Conclusions

In the present study, a Stirling Regenerator Test Bench (SRTB) is designed and constructed for Stirling engine regenerator characterization and a preliminary numerical procedure is presented to characterize pressure drop through the wound woven wire mesh

regenerator matrices. The SRTB is designed for oscillating flow analysis in accordance with the configuration used by Tanaka et al. [6] and the operating range of the micro-CHP Stirling engine WhisperGen™.

At the initial stage of the present study, the relevant geometrical parameters, physical properties of the regenerator material and of the working gas, and operating variables are identified. With regard to pressure drop and convective heat transfer testing, the influencing parameters and variables are included in a functional relationship to be used as basis for obtaining experimental friction factor and Stanton number correlations. Some drawbacks are encountered during the design and setting-up of the SRTB, i.e. the oscillating frequency limitation, the mass flow rate measurements, the regenerator matrix temperature, etc. Therefore, some modification and methods are proposed during the project.

The temperature of the regenerator matrix substantially coincides with the time average of the gas temperature at the corresponding end. Consequently, it seems that temperature measurements of the regenerator matrix are not necessary, which significantly reduces the cost of the tests associated with the correct placement of the thermocouples.

The concept of regenerator thermal efficiency is interesting to carry out comparative tests with different regenerators. The first tests evidence that the efficiency increases with increasing mass flow rate, and that the efficiency is higher for the cooling process than for the heating process in all cases.

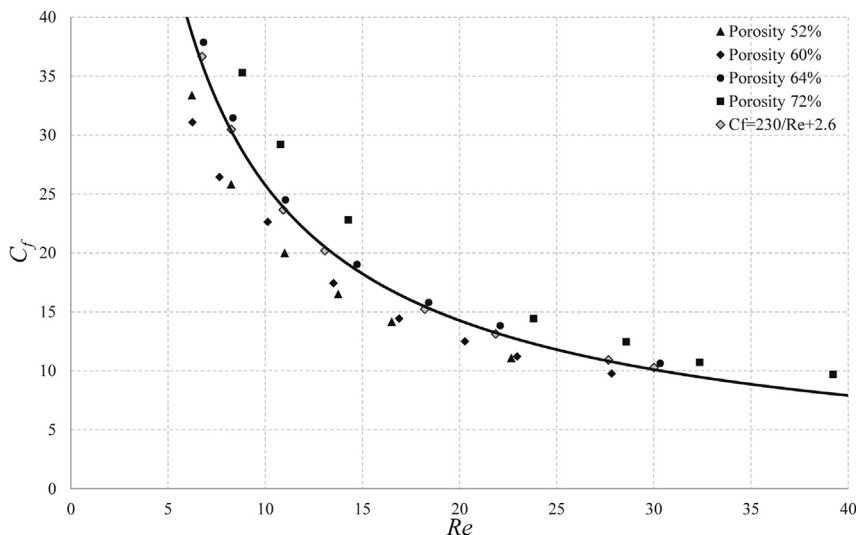


Fig. 17. Darcy friction factor versus Reynolds number.

In the numerical computations on the other hand detailed wire mesh matrix models, i.e. the whole Stirling mesh matrix models are generated with different level of volumetric porosity using an equivalent porous media model. The initial numerical results underline the importance of obtaining well-characterized experimental data for further studies concerning heat transfer phenomena and the use of computational methods in the name of deriving a universal friction factor correlation equation for a proper characterization of Stirling regenerators. Though the proposed numerical methodology reproduces the results which are in good correspondence with the results of Darcy's law, the model validity is required to be further assessed over a very large parametric range and woven wire mesh configurations for future Stirling regenerator applications. The present numerical flow methodology also provides a cost effective alternative to characterize the pressure drop through a woven wire mesh Stirling regenerator obtaining the permeability and inertial coefficient requested in a porous media model for further applications.

Acknowledgments

This work is partially supported by SAIOTEK and the Diputación Foral de Gipuzkoa programs of Basque Government, and Efficient Home Energy (EHE).

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